1939-06

An investigation of some of the characteristics of the magnetic injection valve for diesel engines.

Persons, Henry Stanford

University of California

AN INVESTIGATION OF SOME OF THE CHARACTERISTICS OF THE MAGNETIC INJECTION VALVE FOR DIESEL ENGINES

HENRY STANFORD PERSONS

Thesis
P37

An Investigation of Some of the Characteristics of the Magnetic Injection Valve for Diesel Engines

by

Henry Stanford Persons
Grad. (United States Naval Academy) 1929

THESIS

Submitted in partial satisfaction of the requirements for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

in the

GRADUATE DIVISION

of the

UNIVERSITY OF CALIFORNIA

June 1939

INDEX

LIST OF ILLUSTRATIONS

OBJECT OF INVESTIGATION

The object of the investigation was to determine the characteristics of the magnetic fuel injection valve as regards weight of fuel injected per stroke, regularity and reproducibility of spray penetration, and the start of injection and duration of valve opening under various conditions of speed, pressure and condenser charge.

DESCRIPTION OF APPARATUS

The magnetic injection valve is shown in section in Figure 1. It consists of a needle valve stem with a small projecting shoulder brazed to its end, a plunger core which acts as guide for the needle, an armature with twelve crosswise magnetic laminations in it, which surrounds the plunger core, and a laminated pole core through which the electrical winding of 40 turns of No. 20 wire passes. The valve needle, plunger core, and armature pass through a hole drilled in the pole core at right angles to the laminations of the pole core and armature. The armature is free to move axially through a distance of .033 inches, before coming into contact with a stop. There is .005 inches clearance between the end of the armature and the face of the shoulder on the valve stem. Therefore, the armature lifts the needle after .005 inches of travel and the needle has a total lift of .028 inches. The needle is held against its seat by a spring acting against the end of the stem, and the armature is held in place by another spring. When the armature is in place, its magnetic laminations do not line up with the magnetic laminations of the pole core. Thus, when an electrical current flows through the coil, lines of flux are set up in the pole core laminations and the laminations of the armature. These flux lines produce a force tending to reduce the reluctance of the core by moving the armature axially and aligning the armature laminations with those of the pole core, thus lifting the needle. The entire inner portion of the body is filled with oil under pressure (the oil surrounds valve stem, plunger, armature, and springs). The valve stem has an axial passageway milled in its surface to allow the oil to reach the valve face. When the valve lifts, the oil is forced through

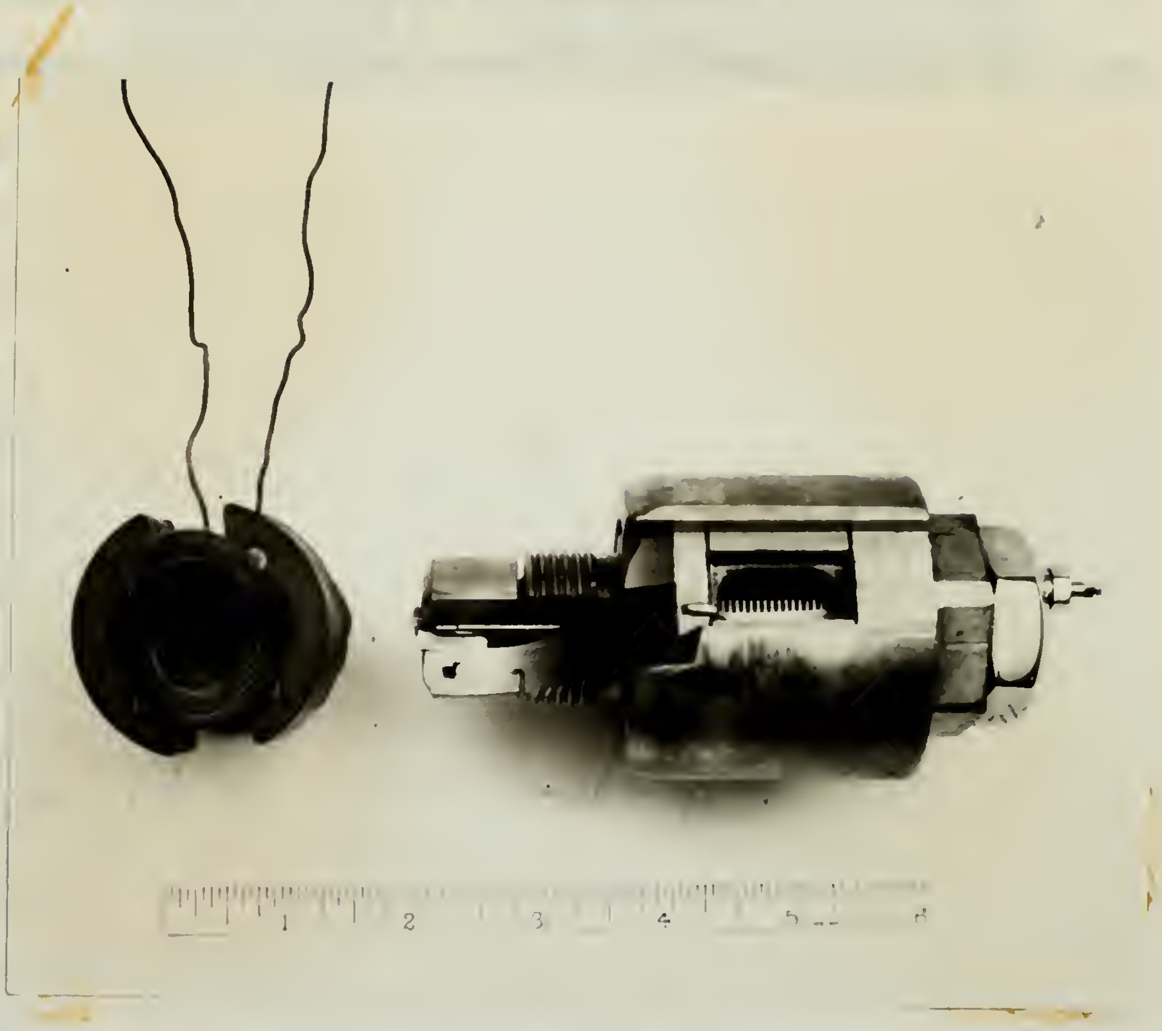


Figure 1

this passageway and thence through an orifice .022 inches in diameter into the cylinder.

The electrical current necessary for operation of the valve is supplied by the discharge of condensers. The condensers are alternately charged by a 24-volt storage battery and discharged through the above mentioned coil by means of breaker points operated by a cam shaft. In the test apparatus there was one condenser of 700-microfarads and three of 200-microfarads capacity, any or all of which could be connected in parallel by means of single pole, single throw, knife-blade switches, as shown in wiring diagram, Figure 3.

The apparatus used in testing the magnetic injection valve is shown in Figure 2. The apparatus consisted of a variable speed motor, A, directly connected to a shaft, on one end of which was mounted the breaker points, B. The breaker points serve to alternately charge condensers, C, from the storage battery, D, and discharge the condensers through the coil of the magnetic injection valve, E. Fuel was supplied to the magnetic valve as follows. A fuel tank, F, was mounted on one platform of a balance to permit measurement of the weight of fuel injected. The balance was fitted with two mercury bath contact switches so that at the instant balance was obtained the circuit was closed and a small neon light, G, was illuminated. Fuel flowed either by gravity or by action of the fuel transfer pump, H, to filter, J, and thence to the high pressure pump, K, (not clearly shown in Figure 2). From the pump the fuel passed, under pressure, to a reservoir, L, and thence to the the magnetic valve which injected it into the spray chamber, M. Another fuel line connected the reservoir with the regulator, N. The regulator was an adjustable, spring-loaded, by-pass valve which allowed the pressure of the fuel supplied to the magnetic valve to be varied.

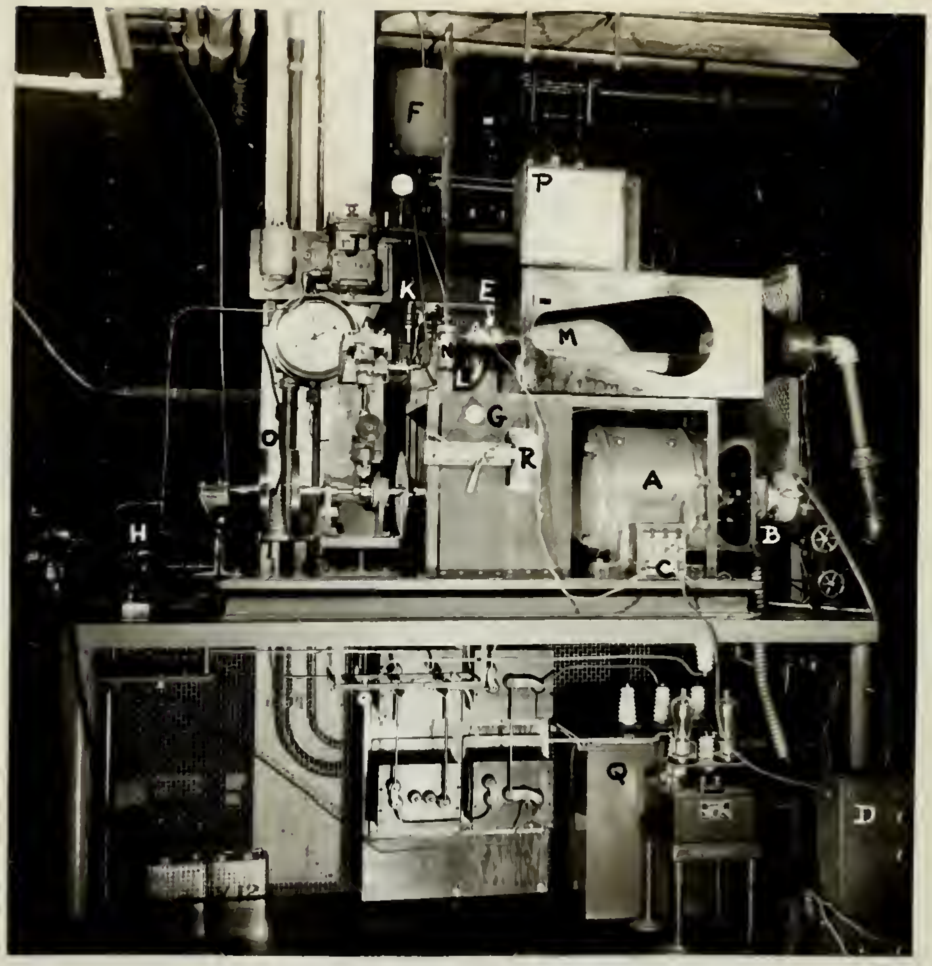


Figure 2

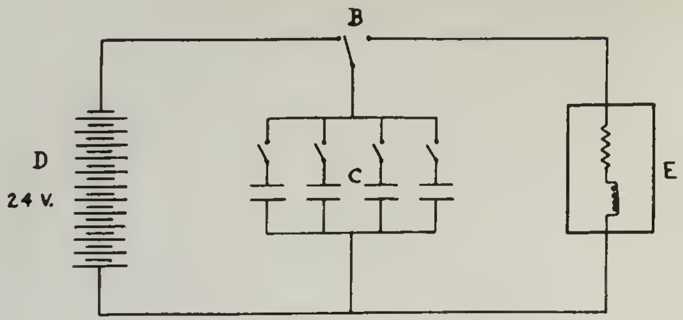


FIGURE 3

SCHEMATIC WIRING DIAGRAM OF THE MAGNETIC INJECTION VALVE

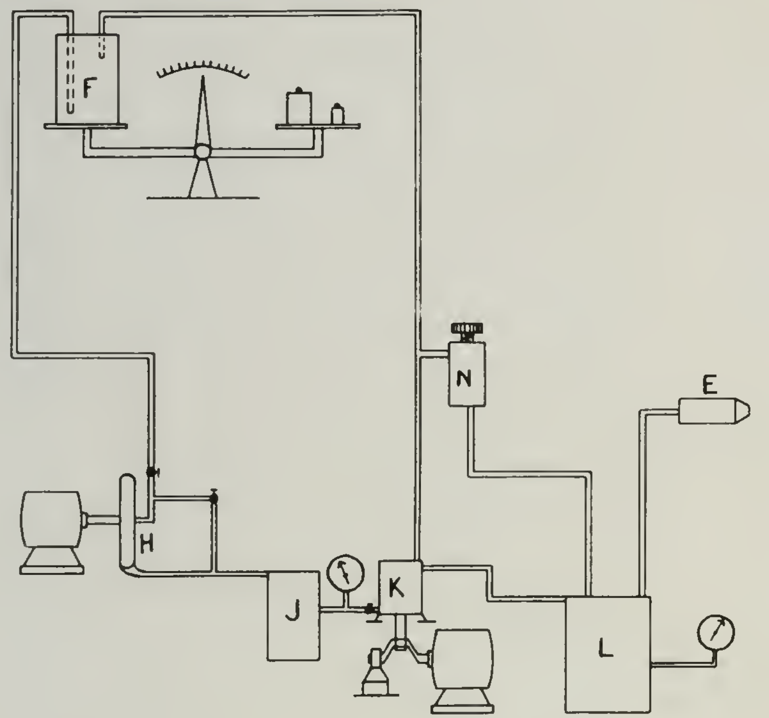


FIGURE 4

SCHEMATIC DIAGRAM OF FUEL SYSTEM

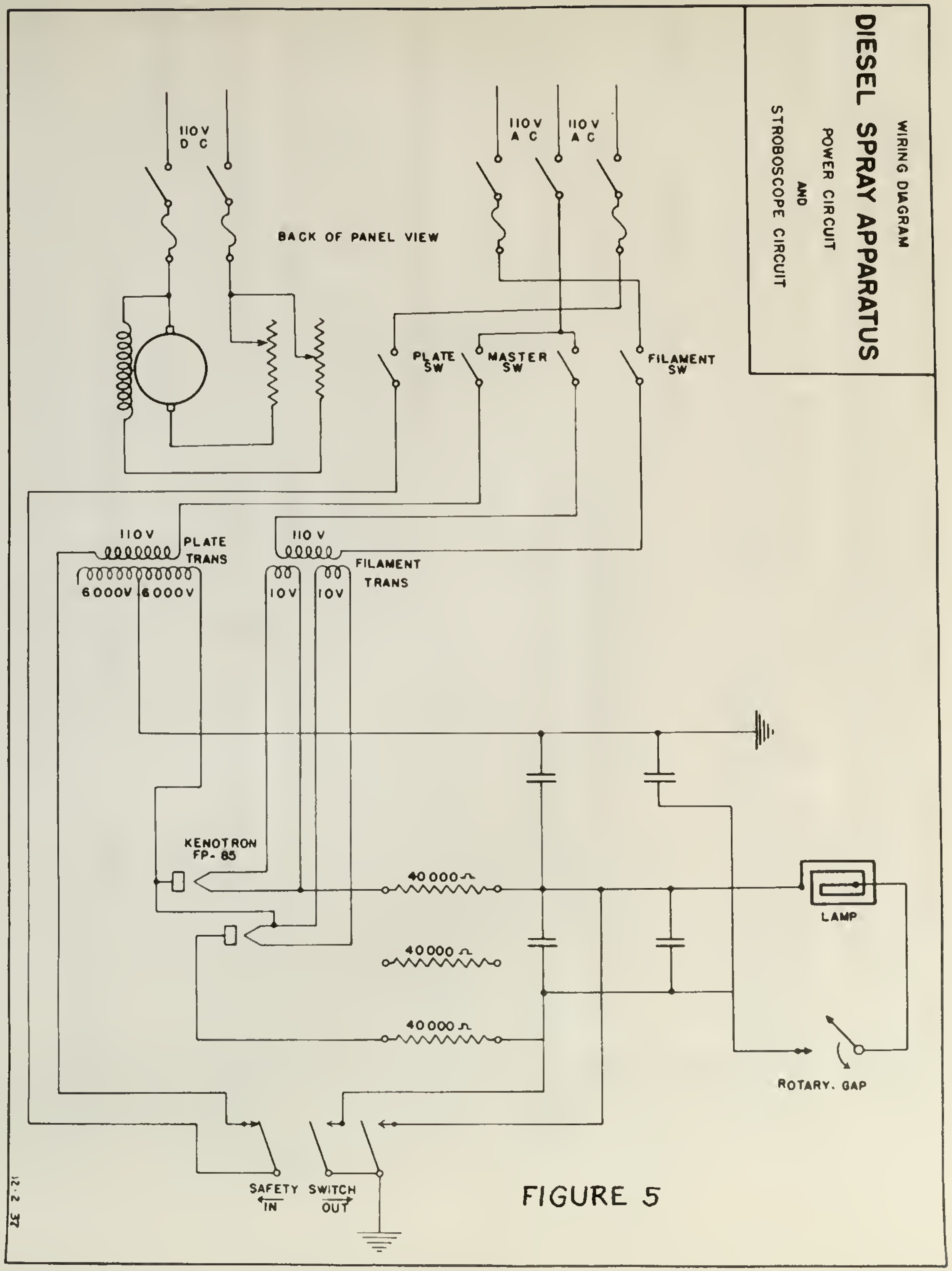


FIGURE 5

This range of variation was from two hundred or three hundred pounds per square inch to thirty-seven hundred pounds per square inch, the maximum capacity of the high pressure pump. The regulator discharged the by-passed fuel to the fuel tank. A schematic diagram of the fuel system is shown in Figure 4.

On the opposite end of the motor shaft from the breaker points was mounted an adjustable, calibrated, rotary spark gap, O, which controlled the instant of illumination of the spray by a neon light, P. The neon light illuminated the spray chamber through an opening in the top. Electrical power for the neon light was supplied by transformers and kenotron tubes, Q. Stroboscopic illumination was thus obtained. The wiring diagram of the power circuit and stroboscopic circuit is shown in Figure 5.

Revolutions were recorded by a counter, R, which was operated through a chain and sprockets from the main shaft.

TEST PROCEDURE

The tests were conducted with oil having the following characteristics:

Gravity (at 71.6°F) degrees API	32.1
Viscosity (at 71.6°F) S.U.S.	41.0
Surface tension, dynes/cm	28.9

In conducting the investigation, the fuel in the system was adjusted to the following pressures, measured at the reservoir: 1000, 1500, 2000, 2500, 3000, and 3700 pounds per square inch; the speeds used were 300, 500, 700, 900, 1100, and 1300 revolutions per minute for each of the above pressures; and at the following condenser capacities for each of the above pressures and speeds: 1300, 1100, 900, 700, and 600 microfarads. In addition to the above, additional tests were run at 2500 lbs per square inch pressure and all the above speeds for condenser capacities of 1460, 1760, 1960, 2160, and 2360 microfarads.

The method of running a test was as follows: the regulator was adjusted to give the desired pressure with the desired condenser capacity and at the desired speed, then the weights on the balance were adjusted until the fuel platform was slightly heavy. When sufficient fuel had been injected to cause the scales to balance, as shown by the small neon light, G, the revolution counter was engaged and a stop watch started. A weight of 0.1 pound, or 0.2 pounds, was removed from the platform balance at this time. Speed was kept constant throughout the run by means of the motor field rheostat and a tachometer. When the scales balanced again, as shown by the light, the revolution counter was disengaged and the stop watch stopped.

The pressure, speed, and condenser capacity were kept constant and the rotary spark gap adjusted until the fuel was seen to be just starting to emerge from the valve orifice. The dial (O, Fig. 2), calibrated in degrees, was read, and this point recorded. The spark gap (O, Fig. 2) was adjusted to a position one degree later and the penetration of the tip of the spray cone read on the scale in the spray chamber. Readings of penetration were made at one degree intervals until the end of injection, as shown by the breaking of the spray cone at the nozzle.

The above procedure was repeated for all combinations of pressure, speed, and condenser capacity stated above.

It should be noted that in all cases where speed is referred to in this paper, camshaft speed (injections per minute) is meant, and not engine crankshaft speed.

DISCUSSION

In general, it is found that fuel injection systems for solid injection Diesel engines may be divided into two general types: the common rail system and the jerk-pump system. The common rail system consists of a reservoir which serves as a common supply source for all cylinders and from which separate fuel lines lead to the injection valves of the several cylinders. The injection valves in this system are actuated by some positive method (cams, etc.)and the fuel in the reservoir is kept at the designed pressure by a single pump. The capacity of the reservoir is very large in comparison to the amount of fuel injected. Pressure surges set up in the fuel line as each valve opens are quickly damped out and are of negligible effect, as is shown by the DeJuhasz analysis given in Figure 10 (a and b)and Appendix I.[(1)*] The magnetic injection valve system is of this type, the electric operation of the valve serving to give a positive, practically instantaneous, opening and closing of the needle valve as contrasted with the slower action of the cam-actuated valve.

The jerk-pump system has a separate high pressure pump for each cylinder, each with its direct lead to the injection valve it supplies. The injection valve is spring loaded and opens when the fuel pressure built up by the jerk-pump in the line exceeds the spring pressure of the valve. The pump discharges fuel into the line to the valve during only a small portion of the stroke, and the fuel discharged from the pump is by-passed to the supply system during the remainder of the stroke. The

*Numbers in parentheses refer to references given in the Bibliography.

interval during which the pump discharges into the fuel line to the valve is controlled by the operator. Investigations[2] have shown quite definitely that there are pressure surges in these systems of such magnitude as to cause the valve to open and shut repeatedly during the desired period of injection with consequent irregularities in the amount of fuel injected and in spray penetration. In spite of these disadvantages, however, the jerk-pump system is in wide use on small engines.

In analyzing the performance of the magnetic injection valve, it is found that three variables; speed, pressure, and condenser capacity, control its performance. In the laboratory all these may be varied by the operator, but in actual practice the injection pressure for an engine is a constant, the speed varies with the load, and the condenser charge is the only variable which may be controlled by the operator. When the effect of these three variables upon: (A), quantity of fuel injected; (B), point at which injection starts; (C), duration of injection; and (D), penetration, was considered, the following results were found.

A. - Quantity of fuel injected. - If the magnetic injection valve is considered as a simple orifice, open only during a small portion of each stroke, and it is considered that the opening and closing of the valve is instantaneous, it may be expected that the weight of fuel injected per stroke will vary with the pressure according to the efflux equation:

$$W = C A \gamma \sqrt{\frac{2g \times 144(p-p_c)}{\gamma}} \text{ lbs/second, where}$$

A = area of the orifice in square feet

γ = weight density of the fuel, pounds per cubic foot

C = discharge coefficient

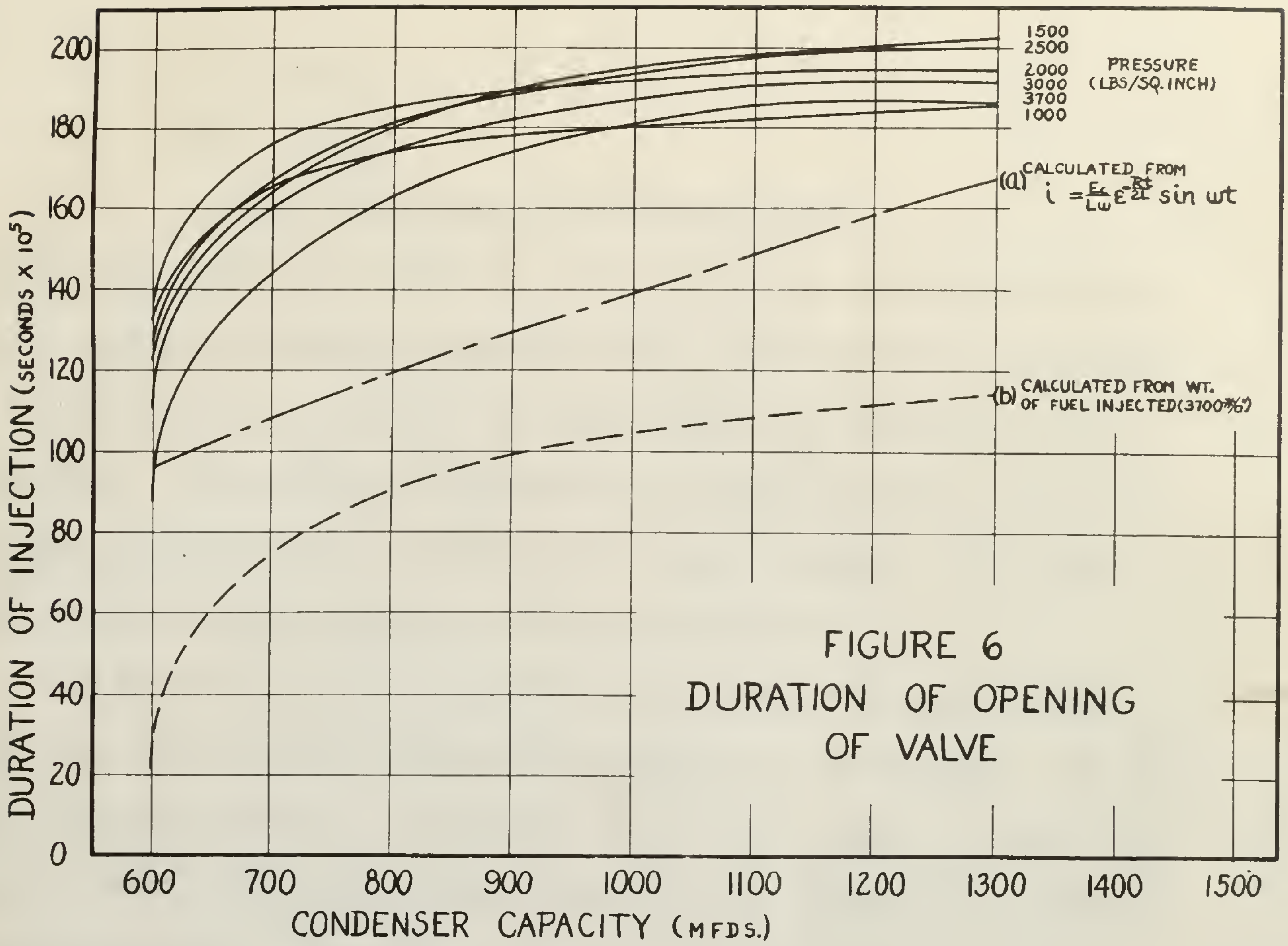


FIGURE 6
DURATION OF OPENING OF VALVE

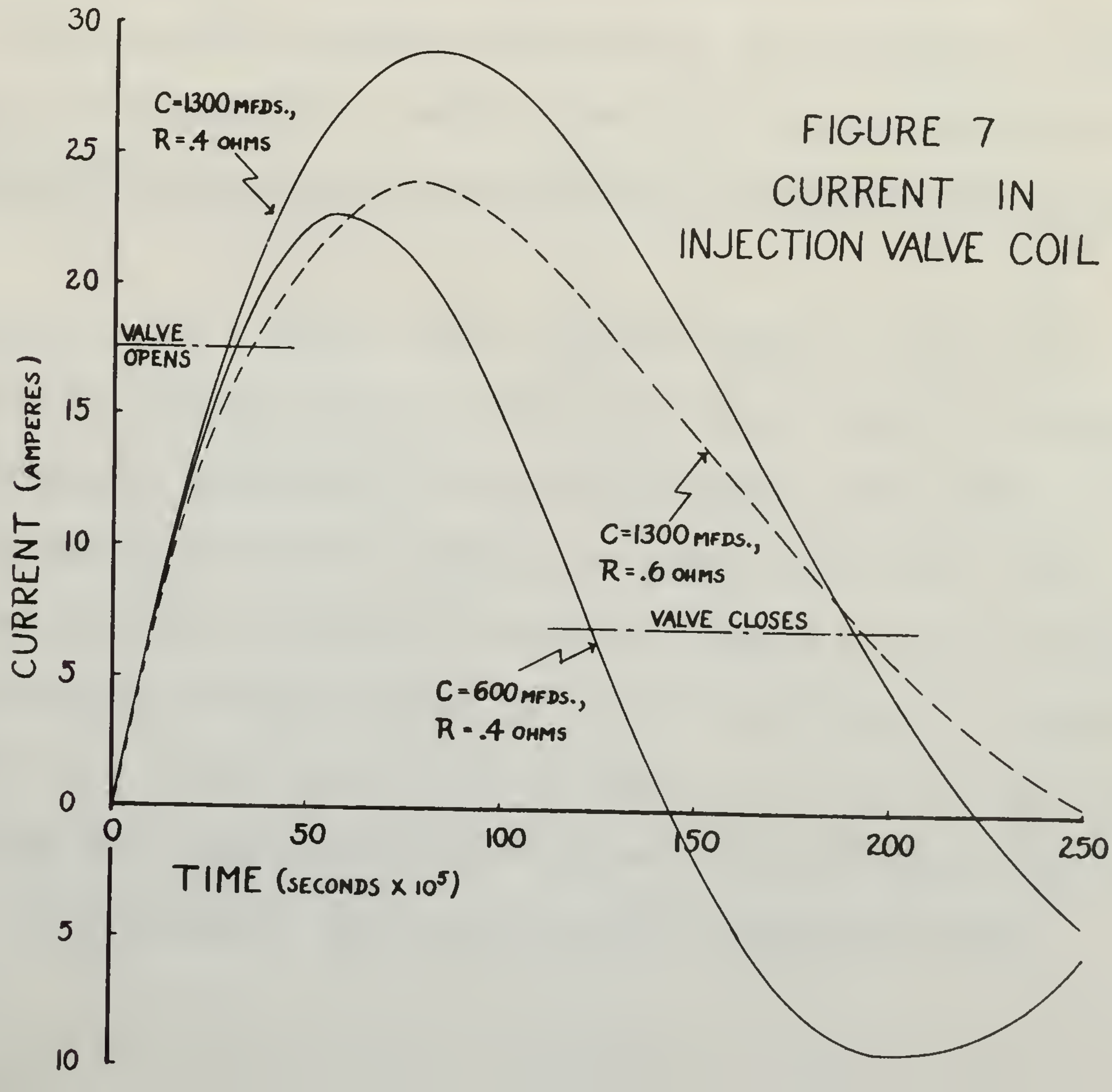


FIGURE 7
CURRENT IN INJECTION VALVE COIL

$R = .4$ ohms

$E_c = 24$ volts

$C = 1300$ microfarads and 600 microfarads

An inspection of Figure 7 will show that for the same circuit resistance, but for different condenser capacities (1300 and 600 microfarads in Figure 7), the point of opening of the valve varies but little with change in capacity. The variation in duration of opening (as shown in Figure 6) is almost entirely due to a change in the time of closing. The valve closes earlier as the condenser capacity is decreased.

The dotted curve ($C = 1300$ mfds, $R = .6$ ohms) in Figure 7 demonstrates the effect of increasing the resistance in the circuit. This increase in resistance has a negligible effect on the points of opening and closing of the valve, but it serves to damp out the oscillating current. This characteristic could be put to great advantage in the case of a circuit so tuned that its succeeding oscillations, after the first, would be so large as to again lift the needle. The use of an increased resistance would be effective in preventing more than one injection per suction stroke.

In general, the quantity of fuel injected varies as predicted. Figure 8 (a to f) shows the weight of fuel injected per stroke for various speeds and condenser capacities, at constant pressures, while Figure 9 (a to d) shows the weight of fuel injected per stroke for various speeds and pressures for constant condenser capacities. Figure 8a (3700 lbs/sq. inch pressure) shows a smooth contour, while the other figures of Figure 8 show "valleys" at various points on their surfaces, the most apparent being the "valley" at 1100 injections/minute in Figure 8c (2500 lbs/sq. inch pressure). In attempting to fix a cause for the appearance of these

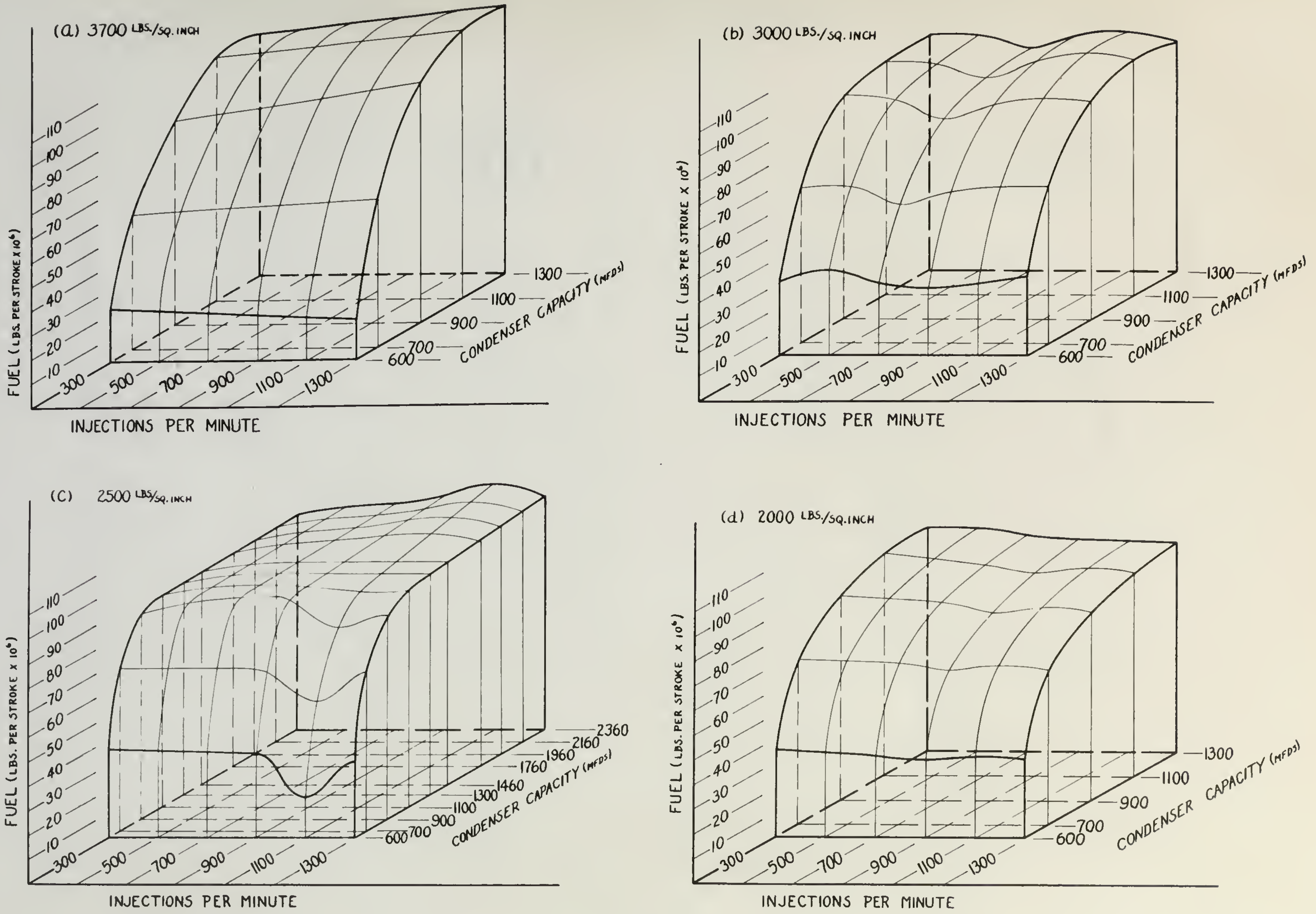
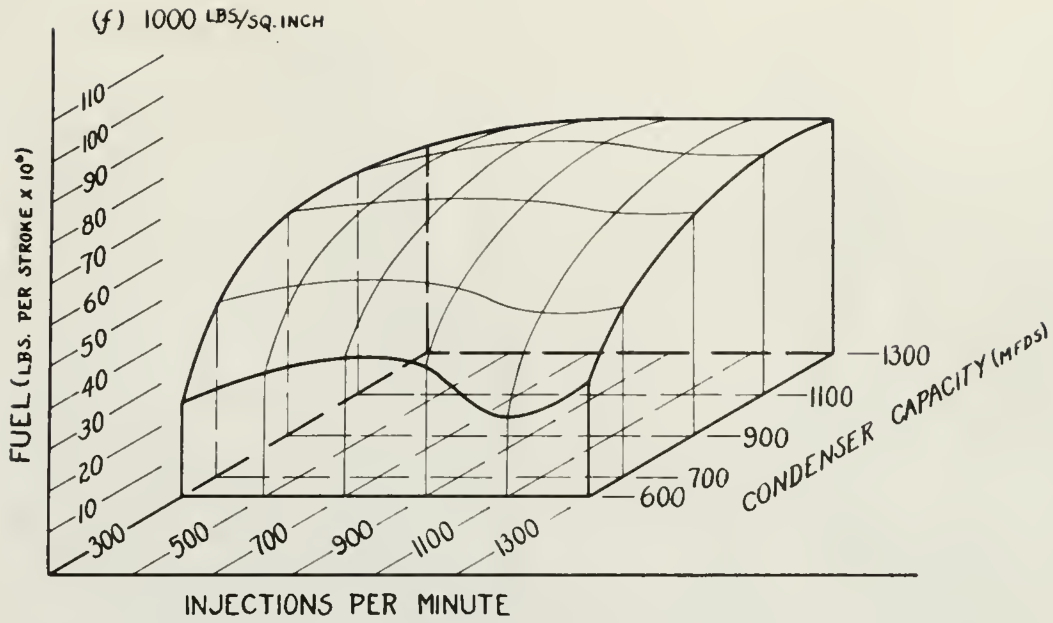


FIGURE 8
THE INFLUENCE OF SPEED AND
CONDENSER CAPACITY UPON FUEL INJECTED



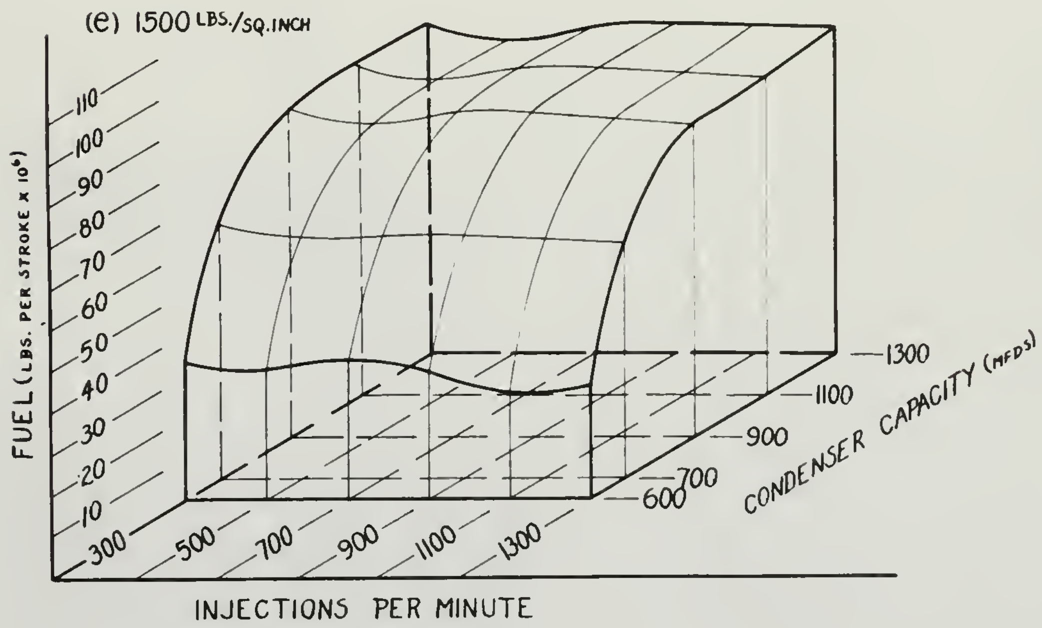


FIGURE 8
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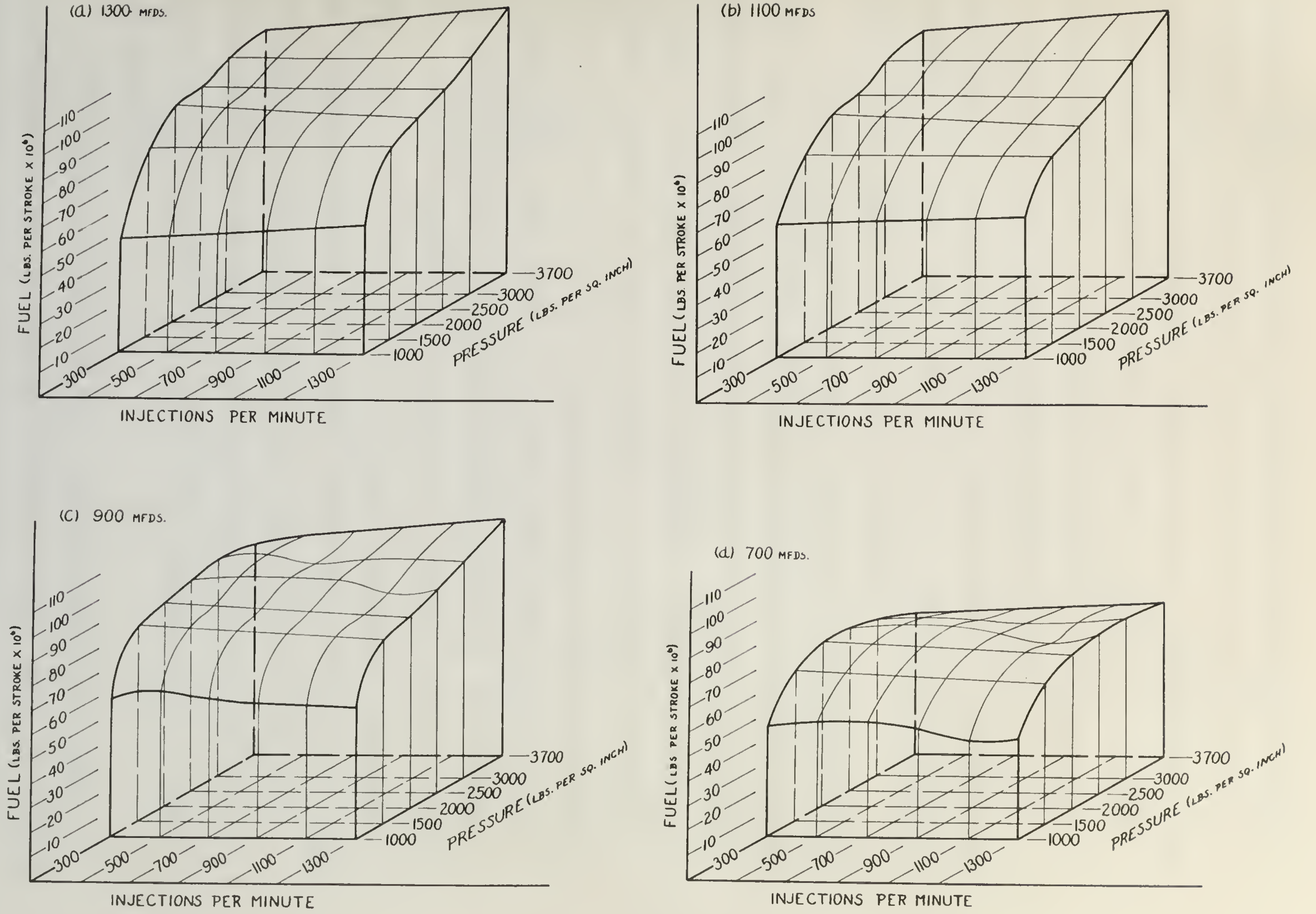

FIGURE 9
THE INFLUENCE OF SPEED AND PRESSURE UPON FUEL INJECTED

"valleys" in the various curves, the following possibilities were investigated: (1) pressure surges in the line between the reservoir and the magnetic valve, momentarily affecting the flow by reducing the value of the pressure term in the efflux equation; (2) presence of possible resonant conditions in the system; (3) the effect of the vibrations of the spring-mass system of the pressure regulator (bypass), N, upon the hydraulics of the system; and (4), the possibility that at low condenser capacities, the force of the magnetic flux was insufficient to keep the valve wide open during the full period of injection.

The effect of possible pressure surges was the first considered, and an investigation was made by means of a graphical analysis developed by De Juhasz[1]. The results of this analysis are shown graphically in Figures 10 (a and b), and the full analysis of the surges at two different pressures is given in Appendix I. An inspection of Figure 10 (b) will show that for the two pressures selected (3700 lbs/sq. inch and 2500 lbs/sq. inch), there is a large pressure drop during the first surge (time equal to 2 L/A) which is rapidly damped out. The magnitude of the pressure variation in the line for the reservoir pressure of 3700 lbs/sq. inch is greater than for the reservoir pressure of 2500 lbs/sq. inch. However, Figure 8a (3700 lbs/sq. inch pressure) shows no "valley", while Figure 8c (2500 lbs/sq. inch) shows a deep valley at this speed (1100 injections per minute). This would seem to indicate that possible pressure surges in the line do not cause these irregularities in weight of fuel injected.

An inspection of Figures 8 and 9 shows no apparent continuity or relationship between the several variables at points where the various "valleys" occur. However, it was found that certain conditions of "valleys" in the various curves, the following possibilities were investigated: (1) pressure surges in the line between the reservoir and the magnetic valve, conceivably affecting the flow by reducing the value of the pressure term in the orifice equation; (2) presence of possible resonant conditions in the system; (3) the effect of the vibrations of the spring-mass system of the pressure regulator (bypass), R, upon the hydraulics of the system; and (4), the possibility that at low condenser capacities, the force of the magnetic flux was insufficient to keep the valve wide open during the full period of injection.

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An inspection of Figures 8 and 9 shows no apparent continuity or relationship between the several variables at points where the various "valleys" occur. However, it was found that certain conditions of

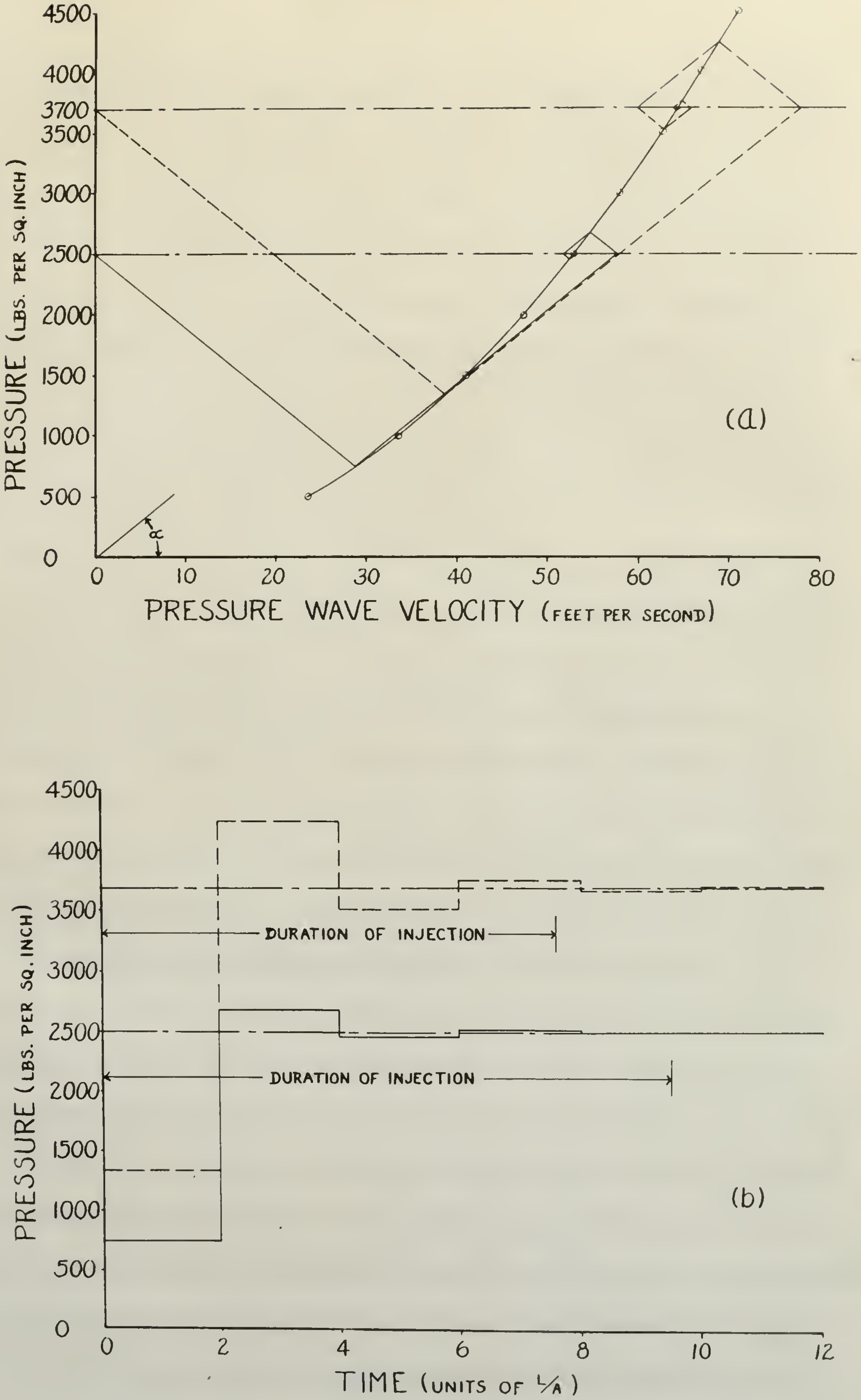


FIGURE 10
ANALYSIS OF PRESSURE SURGES
IN INJECTION VALVE SYSTEM

pressure, speed, and condenser capacity, it was impossible to prevent fluctuations of the pressure in the system, as evidenced by the fluctuations of the needle of the pressure gage. This is believed to have been caused by a resonant condition between the spring-mass system of the regulator, N, and the rest of the system under these conditions of speed and pressure. These fluctuations were about twenty-five lbs/sq. inch above and below the desired pressure. These fluctuations were undoubtedly indicative of pressure variations in the system which, at resonant frequencies, could have attained comparatively large magnitudes. This belief is borne out by the surfaces of Figure 8. Figure 8a (3700 lbs/sq. inch pressure) shows the effect of speed and condenser capacity upon weight of fuel injected at full pump pressure; that is, with the <u>regulator completely closed</u>, and therefore, <u>not in the system</u>. In all other diagrams of Figure 8, the regulator is in use to some degree and irregularities in weight of fuel injected are evident in each figure. Therefore, it is believed that most, if not all, of these irregularities are caused by the regulator. Since the regulator is not a part of the magnetic injection valve system, but simply a laboratory device installed to permit the attaining of various pressures in the analysis of the operation of this valve, its deficiencies are not properly chargeable to the magnetic valve. It is believed that a regulator, installed so as to throttle the supply to the high pressure pump, would be so far removed from the magnetic valve as to obviate any possibility of a recurrence of these effects in a further investigation along the same lines.

Figure 7 (current versus time) indicates that for a condenser capacity of 600 microfarads, the current in the coil of the magnetic valve does not greatly exceed that current necessary to open the valve.

Thus, it is conceivable that there might be occasions when the valve is not opened fully, or although once opened, has a tendency to close early. However, pictures of the spray and visual examination of the spray during the progress of this investigation have failed to disclose any evidence of such intermittent operation at this low capacity. In addition, irregularities are noted in several instances at condenser capacities above 600 microfarads, when the current in the coil is of such value as to preclude any possibility of this factor causing these irregularities. Therefore, it is felt that the small current at the low condenser capacities is not a cause of these irregularities.

In Figure 8 (c), the surface is extended to a condenser capacity of 2360 microfarads. This surface is quite flat and smooth, and indicates that for the valve investigated, any increase of condenser capacity beyond 1300 microfarads is unnecessary.

B. - Point of start of injection. - Theoretically, the point of opening of the valve should not vary with pressure. According to Figure 7 (plot of the current in coil as a function of time) the variation of the time of opening with change in condenser capacity is negligible. The point of opening of the valve varies directly with speed. Table I shows the point of opening of the valve (injection delay), based on 36^{0} (on rotary spark gap dial) as zero point. Within experimental limits, this table shows that there is no change in the injection delay with change in condenser capacity, and that the change in injection delay varies linearly with speed. The change in injection delay with variation in pressure shows that the delay was notably greater for pressures of 1500, 2000, and 2500 lbs per sq. inch than it was for pressures of 1000, 3000, and 3700 lbs per sq. inch. The test runs were made in the following order: 1500, 2000,

2500, 3000, 3700, and 1000 lbs per sq. inch. This fact suggests that some change occurred in the adjustment of the valve or that the spark gap adjustment changed after the 2500 lbs per sq. inch run was completed. The discrepancy in injection delay can be accounted for in no other way.

TABLE I

		Injection Delay (From 36°) in Degrees					
Pressure	Condenser Capacity	Speed 300	500	700	900	1100	1300
1000	1300	.2	1.7	2.2	3.8	6.5	7.5
	1100	.3	1.8	2.2	3.9	6.5	7.2
	900	.3	1.8	2.2	4.0	6.5	8.0
	700	.6	1.9	2.5	3.9	6.8	8.1
	600	.7	2.0	2.8	4.4	7.5	8.7
1500	1300	1.3	3.3	4.3	5.4	8.9	9.0
	1100	1.4	3.2	4.5	5.4	8.9	9.2
	900	1.5	3.6	4.5	5.5	8.9	9.1
	700	1.5	3.8	5.0	5.5	8.8	9.7
	600	1.5	3.8	5.2	6.5	9.2	10.2
2000	1300	1.0	2.5	3.6	6.6	6.7	7.8
	1100	1.2	2.5	3.8	6.3	6.7	8.9
	900	1.2	2.6	4.0	6.3	6.7	9.0
	700	1.3	2.8	4.2	6.1	6.8	9.2
	600	1.4	2.9	4.2	6.2	7.9	10.3
2500	1300	1.2	2.7	3.5	5.0	7.0	7.5
	1100	1.3	2.8	3.5	5.0	7.0	7.6
	900	1.5	2.7	3.7	5.2	7.1	7.7
	700	1.4	2.8	3.7	5.0	7.2	7.7
	600	1.6	3.0	3.9	5.8	7.2	8.0
3000	1300	.1	2.1	3.4	4.9	5.8	6.9
	1100	.2	2.1	3.3	5.0	6.1	6.8
	900	.3	2.1	3.3	5.0	6.2	7.0
	700	.6	2.1	3.6	5.2	6.0	7.5
	600	.7	2.5	3.8	5.2	6.3	7.8
3700	1300	0	2.3	3.6	5.1	6.0	7.2
	1100	0	2.5	3.2	5.1	6.2	8.0
	900	.2	2.5	3.2	5.3	6.3	8.2
	700	.3	2.3	3.2	5.5	6.5	8.2
	600	.6	2.7	3.5	5.4	6.8	8.7

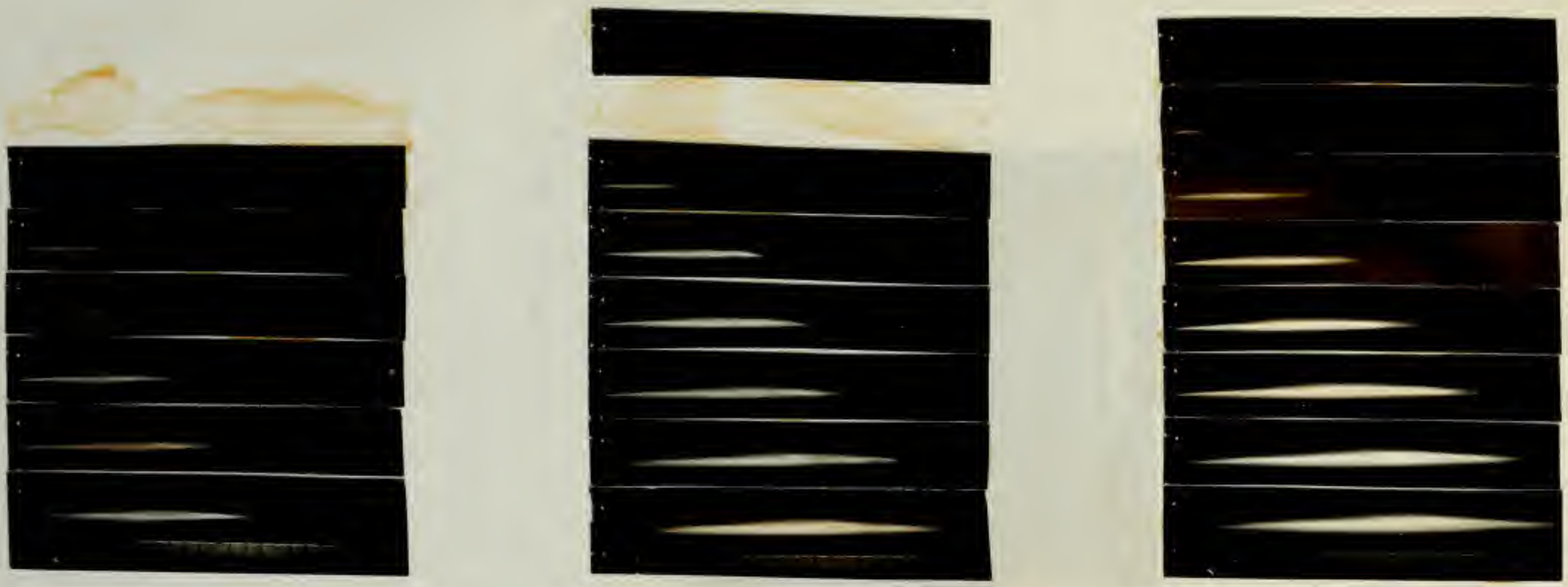
C. - Duration of injection. - Figure 6 shows the variation in the duration of injection (duration of opening) of the magnetic valve with change in condenser capacity, and at constant pressures. Curves of observed duration are shown for each of the six test pressures. Values used for each curve are the mean of values observed at the several speeds used in the investigation. These curves have the same general shape for all pressures and it is believed that the variation in duration of opening at the different pressures is due to the effect of the pressure gradient in the oil (due to flow) acting on the button which is brazed to the valve stem. This pressure difference is greater at the higher values and therefore closes the valve more quickly, thus shortening the duration of opening.

Two other curves are shown in Figure 6, one (a) being a curve made up of the calculated duration of opening based on the current equation (Appendix I), and one (b) being calculated from observed values of weight of oil injected at a pressure of 3700 lbs/sq. inch. In calculating the values for the latter curve, a coefficient of discharge of 0.8 is used, although this coefficient may vary between 0.65 and 0.82[3 and 4]. If a lower coefficient of discharge were used, instead of 0.8, the resulting curve, while retaining the shape and slope of (b) as plotted, would more nearly approximate the height of (a). Comparing curve (b) ($c = .8$) with the curve of observed values (3700 lbs/sq. inch) it is found that the two are similar in shape and slope. Furthermore, curve (b) shows a duration of injection of about 75×10^{-5} seconds less than the observed curve. The observed values are, in reality, the time from the beginning of oil flow as observed at the valve tip until the end of that flow is noted. Therefore, this observed time is in error by the extra time required for the

C. - Duration of Injection. - Figure 5 shows the variation in the duration of injection (duration of opening) of the magnetic valve with change in condenser capacity and at constant pressures. Curves of observed duration are shown for each of the six test pressures. Values used for each curve are the mean of values observed at the several speeds used in the investigation. These curves have the same general shape for all pressures and it is believed that the variation in duration of opening at the different pressures is due to the effect of the pressure gradient in the oil (due to flow) acting on the button which is brazed to the valve stem. This pressure difference is greater at the higher values and therefore closes the valve more rapidly, thus shortening the duration of opening.

Two other curves are shown in Figure 6, one (a) being a curve made up of the calculated duration of opening based on the current equation (Appendix I), and one (b) being calculated from observed values of weight of oil injected at a pressure of 5000 lb/sq. inch. In calculating the values for the latter curve, a coefficient of discharge of 0.9 is used, although this coefficient may vary between 0.80 and 0.82[2 and 4]. If a lower coefficient of discharge were used, instead of 0.9, the resulting curve, while retaining the shape and slope of (b) as plotted, would more nearly approximate the height of (a). Comparing curve (b) (c = .9) with the curve of observed values (5000 lb/sq. inch) it is found that the two are similar in shape and slope. Furthermore, curve (b) shows a duration of injection of about 75×10^{-5} seconds less than the observed curve. The observed values are, in reality, the time from the beginning of oil flow as observed at the valve tip until the end of that flow is noted. Therefore, this observed time is in error by the delay time required for the last particles to flow from the valve seat to the tip. These last particles flow at a reduced velocity since their momentum is the only force propelling them; the pressure differential, which causes the flow, having become zero at the instant the needle valve closed. It is believed that curve (b) is very nearly the curve of actual duration of opening of the valve.

Curve (a) has a minimum value of 96×10^{-5} seconds at 600 microfarads and a maximum value of 167×10^{-5} seconds at 1300 microfarads. The curve closely approaches a straight line, but is slightly convex upwards. Curve (b) has a minimum value of 21×10^{-5} seconds at 600 microfarads and a maximum value of 114×10^{-5} seconds at 1300 microfarads. In the interval from 600 to 700 microfarads, the slope is quite steep, but from 700 to 1300 microfarads the slope decreases gradually until it is quite small at 1300 microfarads. This deviation of actual values from those values predicted by the current flow is due to the inertia of the valve stem, armature, and springs, and to the force of the friction of the oil passing these parts. When the valve is closed, the force exerted by the flux lines is the only force tending to open the valve, while the spring force and the inertia of the movable parts of the valve tend to oppose the opening. The inertia thus tends to make the valve open later than it would otherwise. This effect is most pronounced at the lowest condenser capacity, as Figure 7 shows, for the force available to open the valve at 600 microfarads is only slightly in excess of the force required. At the higher capacities the available force is much greater than that required. After the valve is opened the forces tending to close it are those due to: (1), compression of the spring, (2), the pressure gradient in the oil due to flow acting on the button,



(a) 1500 lbs/square inch

(b) 2500 lbs/square inch

(c) 3500 lbs/square inch

600 rpm, 1100 mfds, pictures at 1 degree intervals

Figure 11

1300 mfds

1100 mfds

900 mfds

700 mfds

600 mfds

2500 lbs/square inch, 600 injections per minute

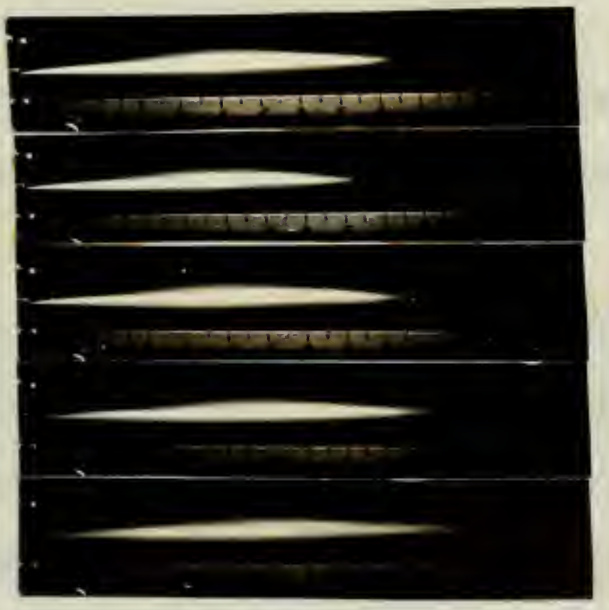
Pictures taken at cut-off

Figure 12

600 microfarads (4 inches). Figure 13 shows pictures taken with valve operating at 2500 lbs/sq. inch, 1100 microfarads, at a time of .002 seconds after start of injection, and at various speeds. The pictures show the spray for speeds of (from top to bottom): 1200, 1000, 800, 600, and 400 injections per minute. The penetration at first glance appears slightly irregular, but when it is considered that at 1000 rpm (the speed at which the greatest irregularity occurs in the pictures), a difference in setting of one degree means about 2½ inches difference in penetration, the deviation from the average of about 1½ inches at these speeds is easily explained. The vibration of the apparatus (at higher speeds) between the time of setting the spark gap and of taking the picture could have caused this change. In addition, any variation in the printing of the picture tends to change the amount of the feathery tip of the spray that is shown and thus change the penetration slightly.

All the pictures of the spray in Figures 12, 13, and 14 are time exposures and therefore are not pictures of single sprays, but of several sprays.

In the practical use of this magnetic injection valve, a condenser of constant capacity will be used and the charge on the condenser, and therefore the weight of fuel injected per stroke, varied by means of a variable resistance in series in the battery circuit. Thus, a more flexible control of the weight of fuel injected can be obtained by this method than by the method used in this investigation.



1200 rpm

1000 rpm

800 rpm

600 rpm

400 rpm

2500 lbs/square inch, 1100 mfds, picture taken .002 seconds after start of injection.

Figure 13

CONCLUSIONS

The results of this investigation lead to the following conclusions:

1. - That, for a constant pressure, the weight of fuel injected per stroke is a function of condenser capacity, and to a very much lesser extent, of speed;

2. - That, for the magnetic injection valve investigated, the practical range of condenser capacities is from 600 to 1300 microfarads; and

3. - That, because of the regularity and reproducibility of the spray, an engine equipped with magnetic injection valves should be smoother running and more economical than an engine equipped with jerk-pumps.

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ACKNOWLEDGEMENT

The author desires to express his grateful acknowledgement to the following:

Professor Carl J. Vogt, for his helpful criticism and suggestions throughout the progress of this work;

The Atlas Imperial Diesel Engine Company, for the magnetic injection valve used in this investigation; and

The W. P. A., for the construction of the apparatus with which the investigation was made.

APPENDIX I

Analysis of Pressure Surges in Fuel Injection Line

Reference: "Hydraulic Phenomena in Fuel Injection Systems for Diesel Engines," K. J. DeJuhasz, Transactions, A.S.M.E., November, 1937, pages 669-670.

Assume a pressure of 2500 lbs/sq. inch, a speed of 1100 injections per minute, and a condenser capacity of 600 microfarads

Let acoustical velocity of oil, $a = 4700$ ft/sec

modulus of elasticity of oil, $k = 284{,}000$ lbs/in^2

coefficient of discharge of nozzle, $\mu = .6$

specific gravity of oil, $\gamma = .8644$

fuel line length, $L = 7$ inches $= \frac{7}{12}$ ft

internal diameter of fuel line, $D = .06$ inches

magnetic injection valve orifice diameter, $d = .022$ inches

$$\tan\alpha = \frac{k}{a} = \frac{284{,}000}{4700} = 60.4$$

$$\frac{2L}{a} = \frac{2 \times 7/12}{4700} = .000248 \text{ seconds}$$

At a pressure of 2500 lbs/sq. inch, a condenser capacity of 600 microfarads, and 1100 injections/minute, period of injection is 6.8° or .00118 seconds

$$V_n = \frac{f}{F}\mu\sqrt{\frac{2g \times 144}{\gamma \times 62.4}}\sqrt{p - pc} = \frac{f}{F} A \sqrt{p - pc}$$

$pc = 0$ lbs/sq. inch gage

$$A = \mu\sqrt{\frac{2g \times 144}{\gamma \times 62.4}} = .6\sqrt{\frac{2g \times 144}{.8644 \times 62.4}} = 7.87$$

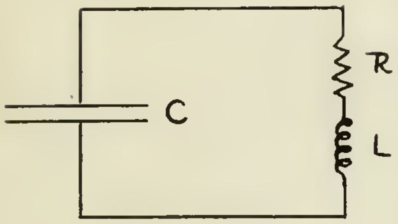
$$\frac{f}{F} = \frac{\pi/4(.022)^2}{\pi/4\,(.06)^2} = .134$$

$$V_n = .134 \times 7.87\sqrt{p} = 1.057\sqrt{p}$$

p (lbs/sq. inch)	V_n (ft/sec)
500	23.6
1000	33.4
1500	40.9
2000	47.25
2500	52.85
3000	57.9
3500	62.5
4000	66.9
4500	70.9

APPENDIX II.

Reference - "Principles of Alternating Currents" - Lawrence, pages 164, 165.



$$i = \frac{E_c}{L\omega} \varepsilon^{-\frac{Rt}{2L}} \sin \omega t, \text{ where}$$

$$\omega = \sqrt{\frac{1}{L\,c} - \frac{R^2}{4L^2}}$$

$R = .4$ ohms $L = 3.25 \times 10^{-4}$ henries $C = 1300$ microfarads

t (seconds)	i (amperes)	t (sec.)	i (amps)	t (sec.)	i (amps.)
.0001	6.92	.0007	28.00	.0015	17.72
.0002	12.90	.0008	28.85	.0017	12.40
.0003	18.10	.0009	28.75	.0019	7.19
.0004	21.95	.0010	27.90	.0022	.49
.0005	24.95	.0011	26.55	.002225	0
.0006	27.10	.0013	22.65	.0025	(-) 4.24

For the above circuit, $\omega = \sqrt{\frac{1}{Lc} - \frac{R^2}{4L^2}} = 1411$

$$\frac{R}{2L} = 615.5$$

$$\frac{E_c}{L\omega} = 52.3$$

$R = .4$ ohms $L = 3.25 \times 10^{-4}$ henries $C = 600$ microfarads

$\omega = 2179$ $\frac{R}{2L} = 615.5$ $\frac{E_c}{L\omega} = 33.9$

t (seconds)	i (amperes)	t (sec.)	i (amps.)	t (sec.)	i (amps.)
.0001	6.89	.0007	21.65	.0015	(-) 1.73
.0003	17.12	.0009	18.03	.0017	(-) 6.36
.0004	20.00	.0010	15.05	.0019	(-) 8.85
.0005	22.05	.0011	11.67	.0022	(-) 8.72
.0006	22.60	.0013	4.60	.0025	(-) 5.36
		.00144	0		

$R = 6$ ohms $\quad L = 3.25 \times 10^{-4}$ henries $\quad C = 1300$ microfarads

$\omega = 1232 \qquad \frac{R}{2L} = 923 \qquad \frac{Ec}{L\omega} = 59.9$

t (seconds)	i (amperes)	t (sec.)	i (amps.)	t (sec.)	i (amps.)
.0001	6.71	.0007	23.75	.0015	14.40
.0003	15.96	.0009	23.35	.0020	5.92
.0005	20.78	.0010	22.40	.002545	0

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P35 Persons
An investigation of some of the characteristics of the magnetic injection valve for diesel engines.